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DEVELOPMENT TESTING OF FREE PLANET TRANSMISSION CONCEPT

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Curtiss-Wright Corporation

Prepared for:

Army Air Mobility Research and Development Laboratory

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DEVELOPMENT TESTING OF FREE PLANET TRANSMISSION CONCEPT

Curtiss-Wright Corporation
One Passaic Street
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June 1975

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Prepared for

EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
Fort Eustis, Va. 23604

EUSTIS DIRECTORATE POSITION STATEMENT

This program is an extension of the work presented in USAAMRDL Technical Report 74-27. This report describes the results of a test program to determine the transmission's behavior under accelerating and decelerating application of load and speed, endurance capability under fifty 1-hour cycles, and the effect of increased gear mesh backlash on the mechanical efficiency of the transmission.

E. Rouzee Givens of the Technology Applications Division served as project engineer for this effort.

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This report presents the results of an experimental program to further demonstrate and evaluate the Curtiss-Wright free planet power transmission concept.

The program consisted of experimentally evaluating the effect of increased gear mesh backlash, cyclic endurance, and transients. Results of the program are as follows:

Block 20. Abstract - continued.

1. Increased Backlash

The .002-inch increase in backlash and revised tip and flank relief to the upper limit of the blueprint resulted in the elimination of back-loading indications on the sun and fixed ring meshes. Markings indicating slight back-loading were still observed on one output mesh. They have been reduced, however, from a full face occurrence to less than .125 inch on one end.

2. Cyclic Endurance

The 50 hours of cyclic endurance testing were successfully completed. During this testing the efficiencies of the reworked system were determined. The result was an increase in efficiency over previous testing.

3. Dynamic Evaluation

The dynamic investigation conducted concurrent with the cyclic endurance testing showed no indication of dynamic problems.

4. Acceleration Evaluation

Twenty-five cycles of snap accelerations were successfully completed with no detrimental effect on the test hardware.

5. Rapid Load Changes

Twenty-five cycles of the duty cycle used for cyclic endurance were successfully completed utilizing load changes accomplished in 2 seconds.

PREFACE

This final report covers continued development demonstration of the Curtiss-Wright free planet power transmission concept. This work includes demonstration of revised test hardware and cyclic testing to simulate application type environments.

The program was conducted during the 8-1/2-month period from 24 April 1974 to 9 January 1975 for the Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, under Contract DAAJ02-74-C-0041, DA Project 1G262207AH89.

Technical direction was provided by Mr. R. Givens of the Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory.

The program was conducted at the Wood-Ridge facility of the Curtiss-Wright Corporation by the Power Systems Department.

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INTRODUCTION

The requirements for transmissions for advanced helicopter and V/STOL aircraft have become increasingly demanding as requirements for lighter weight and greater reliability are continuously accentuated. The work summarized in this report addresses these requirements. This work represents a new concept in power transmission speed reducers and speed increasers. The concept is a derivative of the Curtiss-Wright power hinge design.

Prior to this contractual effort, engineering studies were conducted by the Curtiss-Wright Corporation to determine the feasibility of such a concept from a design point of view and to evaluate and compare this new transmission with currently available transmission systems. These studies have shown that, for example, in applications where multistage conventional planetary designs are normally employed, the new free planet concept offers the following potential advantages and benefits:

1. Improved reliability to the extent that it eliminates conventional planet bearings.

2. Lower weight.

3. Reduced sensitivity to lubrication variations because of elimination of bearings and potential of greater serviceability after loss of lubricant.

4. Fewer parts, less critical tolerances, and lower overall cost.

Previous contractual[1] effort provided the second step of the development cycle, namely, designing, manufacturing, and demonstrating development units. The objective of the program was to verify some of the engineering work completed and to demonstrate the concept with real (load-carrying) hardware. These development units were rated at approximately 500 horsepower with approximately a 20:1 reduction ratio. The characteristics of this new drive system were investigated at loads and speeds compatible with current reduction gear applications.

The current contractual effort reported herein continues this development cycle. In this effort, reworks were provided to the gear teeth to provide increased backlash to the system. The effects of cyclic endurance and accelerations and decelerations were investigated, and attempts were made to visually define the spindle dynamics.

[1] De Bruyne, Neil A., DESIGN AND DEVELOPMENT TESTING OF FREE PLANET TRANSMISSION CONCEPT, Curtiss-Wright Corp., USAAMRDL-TR-74-27, Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, April 1974, AD 782857.

DESCRIPTION OF DEVELOPMENT HARDWARE

The free planet concept has been developed in this program utilizing test hardware designated FP501R. The demonstration hardware provided a reduction ratio of 19.2425. The rating of this gearbox is 500 horsepower with an input speed of 8000 rpm.

The FP500, the basic unit, is a planetary gear assembly which is compounded and consists of three gear meshes. The first plane of the planet gear meshes with the sun gear, the second plane meshes with the output internal ring gear, and the third plane meshes with a stationary internal ring gear. The first and second planes of the planet gear are splined, double piloted, and locked to a quill shaft by a nut and cup lock. The third plane is splined, double piloted, and locked to the second plane of the planet gear by a nut and cup lock. The gears are timed so that the second and third plane planet gears have a tooth in line at the top vertical centerline and the first plane planet gear has a tooth in line 180° at the bottom vertical centerline.

The FP501 is essentially the same as the FP500 except that the quill shaft has been eliminated and torque is transmitted through the hollow support shaft.

Figure 1 shows the basic FP500 layout, and Figure 2 shows the revisions to make the FP501 layout drawing. Each of these configurations utilizes five planet spindles.

The FP501R utilized for this program was essentially the same as the FP501 except for a nominal .002-inch increase in backlash and a change to the involute form to provide .0006-inch tip relief and .0002-inch flank relief. A typical involute rework is shown in Figure 3.

In order to conduct the dynamic investigation, the viewing port shown in Figure 1 was increased to the maximum diameter possible within the welded envelope. This increased the viewing area by approximately 70%.

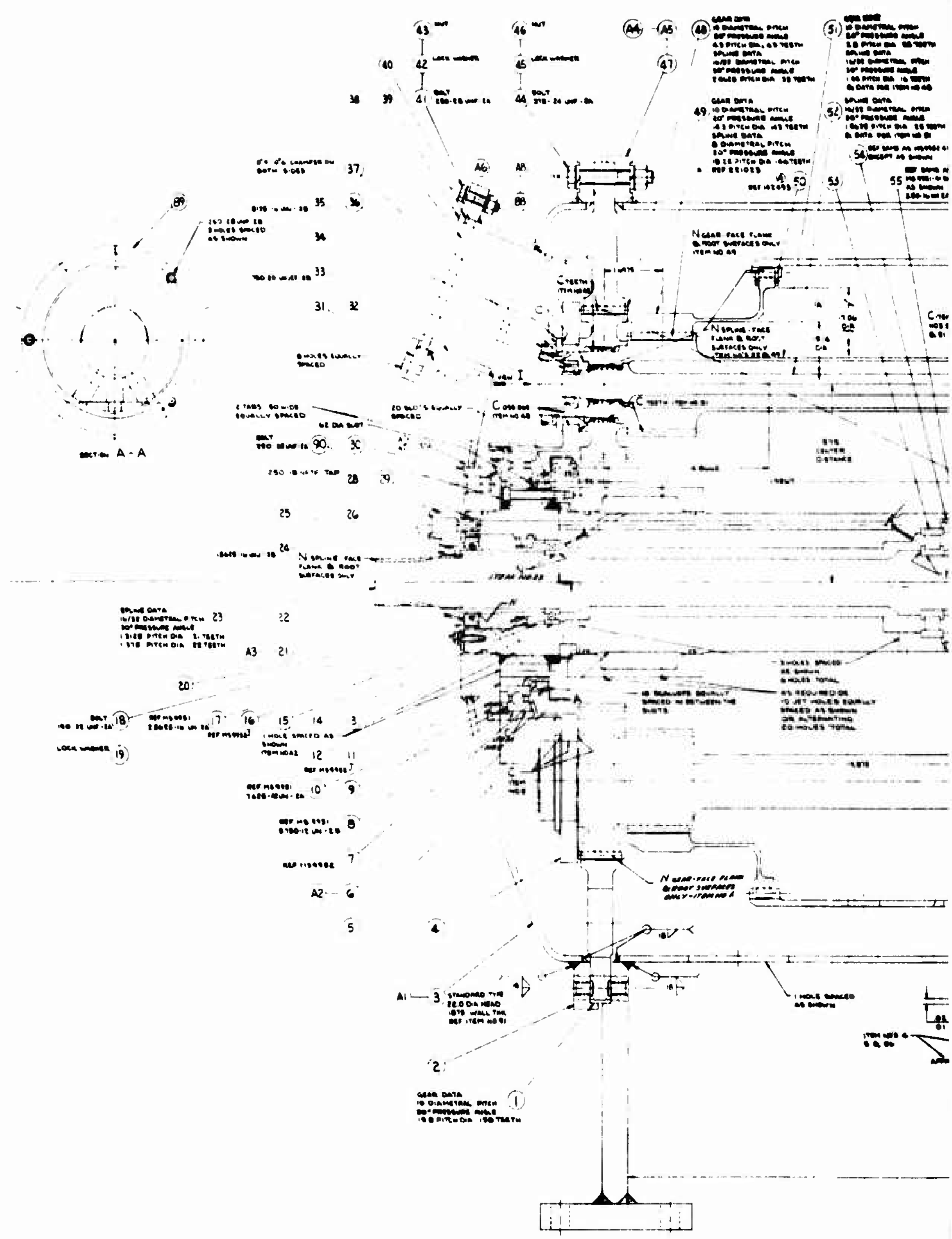
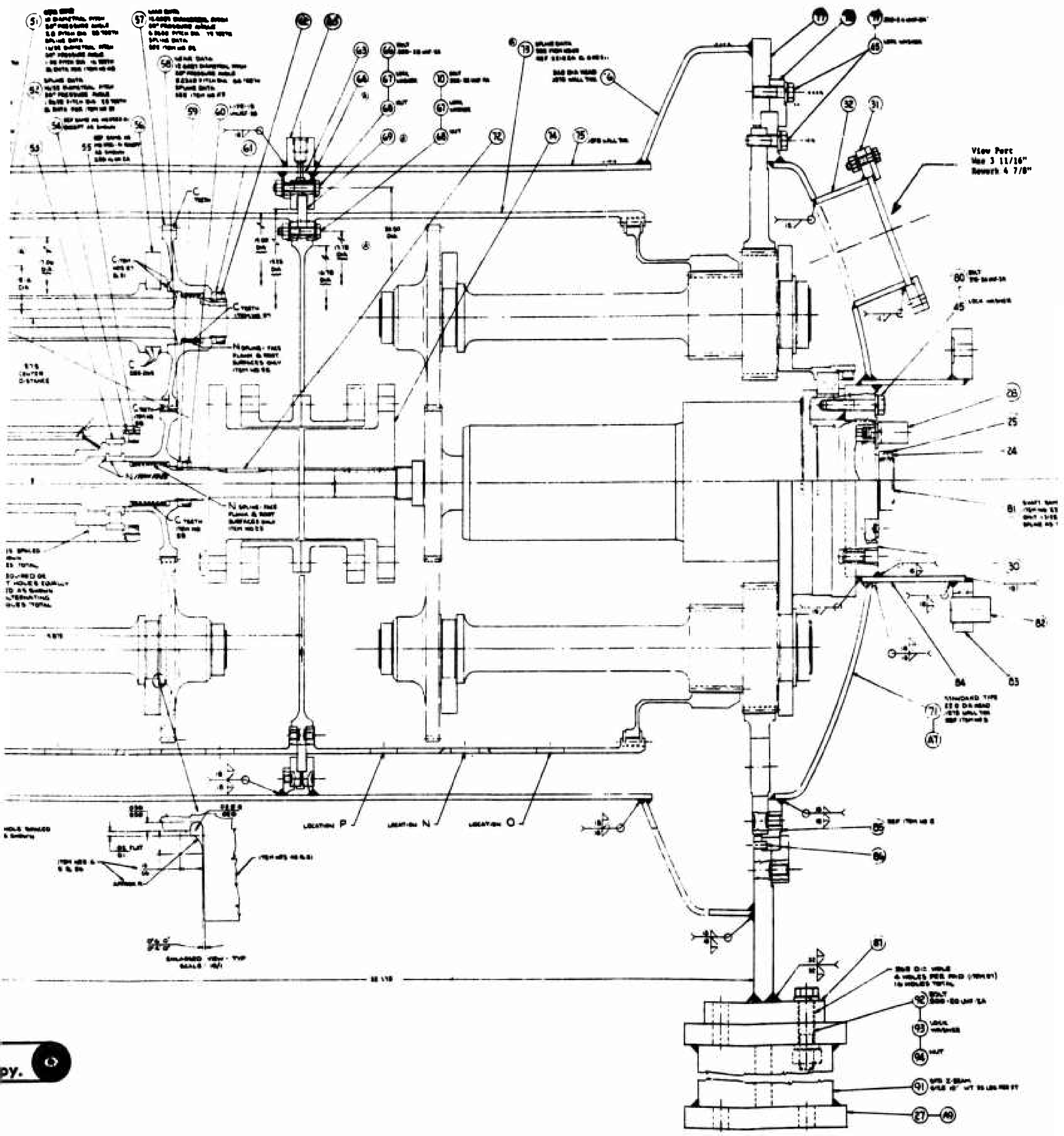


Figure 1. Free Planet Transmission FP500.



View Port
Was 3 11/16"
Rework 4 7/8"
LOCATION P
LOCATION N
LOCATION O

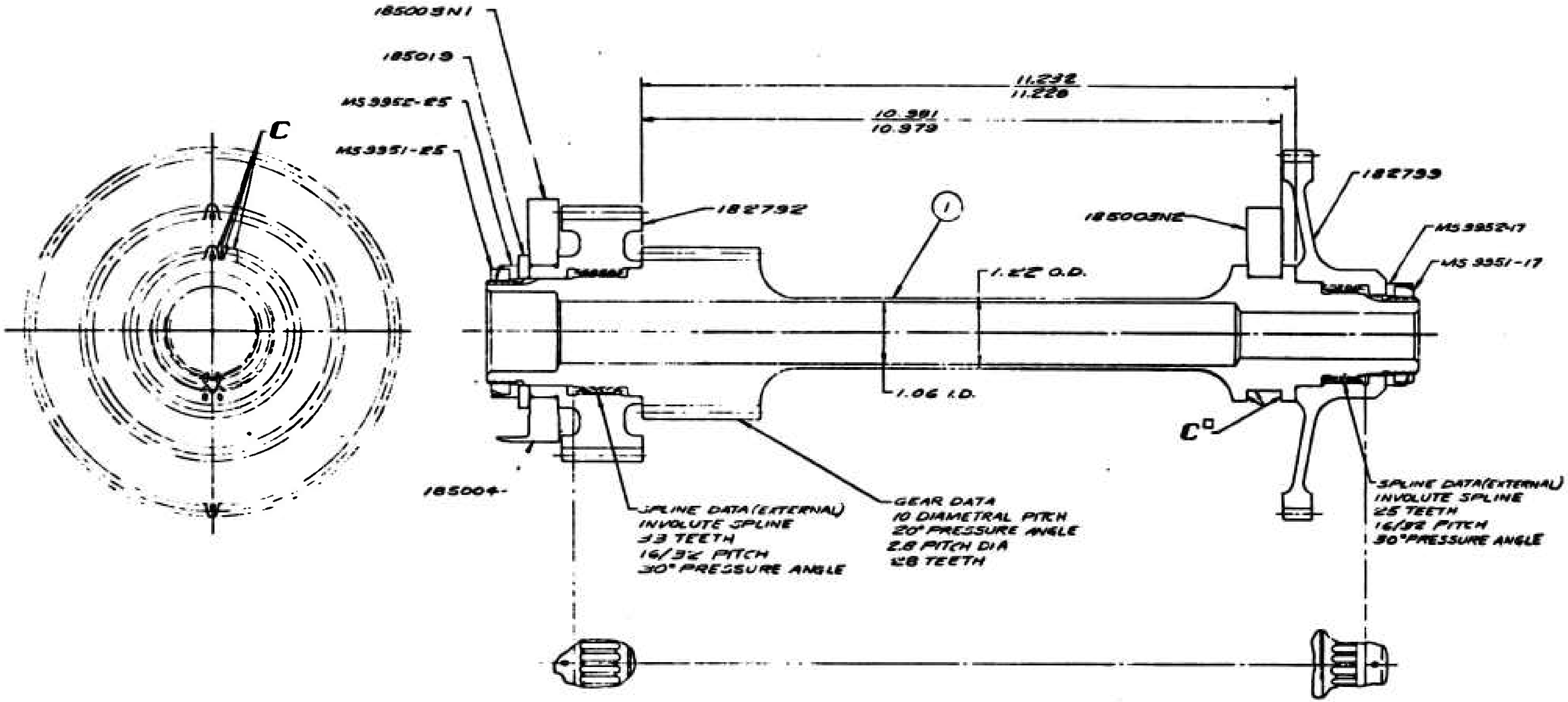


Figure 2. Free Planet Transmission FP501.

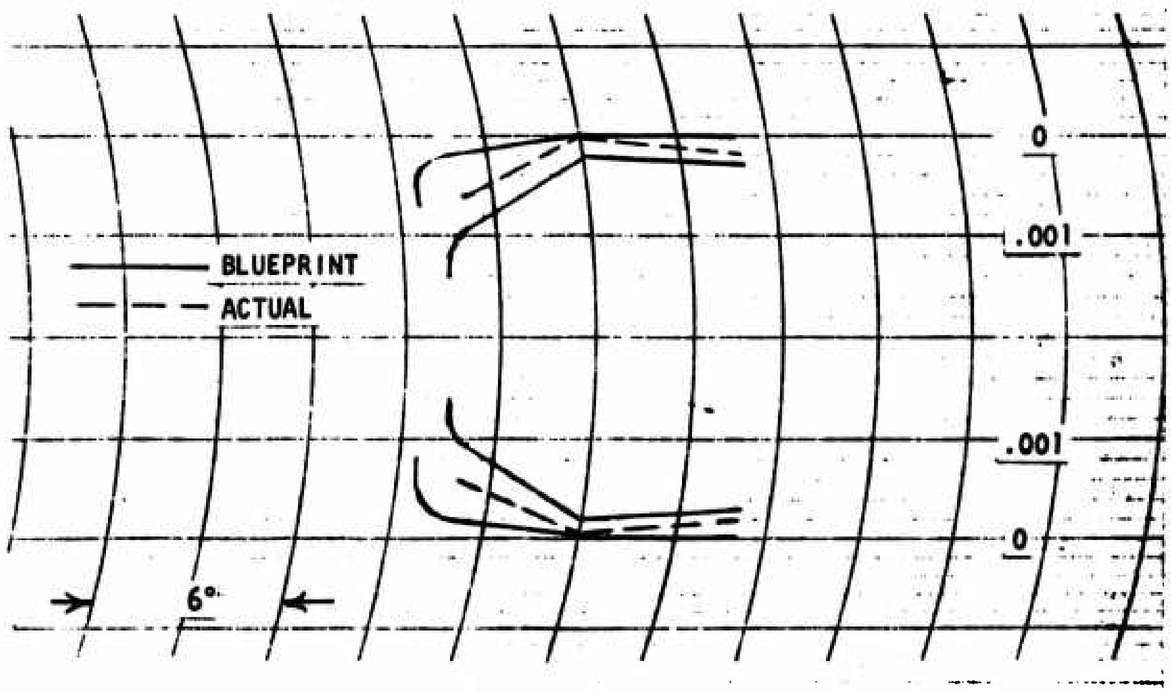


Figure 3. Typical Involute Profile.

TEST PROGRAM

The following experimental evaluation program was undertaken to more fully define the operating characteristics of the Curtiss-Wright free planet concept.

1. Effect of Increased Backlash:

    a. Testing was conducted for approximately 5 hours at rated speed (8000 rpm input) and load (500 hp).

    b. Efficiency was determined.

    c. Effect of operation on visual appearance of test hardware was noted.

2. Cyclic Endurance:

    a. Fifty hours of cyclic endurance testing was conducted as defined by Duty Cycle, item (b).

    b. Duty Cycle

Speed	Torque	Time
8000 rpm	20% of Rated	5 Minutes
8000 rpm	100% of Rated	15 Minutes
8000 rpm	50% of Rated	20 Minutes
8000 rpm	100% of Rated	15 Minutes
8000 rpm	25% of Rated	5 Minutes

    c. During the cyclic endurance test a stroboscope was utilized to view the dynamics of the fixed ring gear mesh.

    d. Motion pictures were taken of operation during cyclic endurance.

3. Acceleration and Deceleration Testing:

    a. Twenty-five acceleration cycles to full speed in 10 seconds at 10% load were made.

    b. Twenty-five cycles utilizing the speeds and loads of the duty cycle item (2b) with changes of load in 2 seconds. Each point was stabilized for a maximum of 5 minutes and a minimum of 2 minutes.

TEST METHOD

A back-to-back or regenerative arrangement was used for the test evaluation of the FP501R transmission. In this configuration the input or sun gear shafts of the two transmissions are fixed to each other by a coupling, with one shaft extending beyond the rig housing to a drive shaft. The output ring gears are also connected to each other by a splined drum. The direct connecting of the ring gears is a simplified form of attaching output shafts to the output ring gears and coupling the shafts. The nonrotating or fixed ring gears attached to the transmission housing absorb the reaction torque. One of the fixed ring gears was rotated by a torque arm to introduce and lock torque into the planetary gearing system. The hydraulic actuator loading the torque arm was calibrated in order to indicate the torque in the planetary gearing.

The test stand control panel is shown in Figure 4. During all testing, constant recordings of the acoustic emission were made by the equipment shown in Figure 5. These recordings were made to aid in diagnosing any problems that could occur during testing. A second view of the test rig mounted on the test stand, showing the torque arm and loading ram, is shown in Figure 6. Schematics of the test stand configuration are shown in Figures 7 and 8.

Back-to-back testing also provides a very accurate means of determining the mechanical efficiency of the system under evaluation. Unlike open-loop testing where torque measuring instrumentation must measure the full torque with its attendant inaccuracy, the back-to-back testing using instrumentation with the same accuracy range as above requires the measurement of only the power loss. For example, a 500-HP transmission in an open-loop system using instrumentation with an overall accuracy of ± 1% would span power levels of 495 to 505 HP, and the observer would not be able to determine the true mechanical efficiency of his system within that range. For accurately made helicopter transmissions where mechanical efficiencies run around 98 to 99%, this type of efficiency measuring approach will not yield satisfactory results. However, in a closed-loop system where the torque component of the rated power is applied through some external mechanism such as a torque arm or hydraulic force, the power required to drive the entire system to the desired speed represents the power loss which can be translated directly into mechanical efficiency of the system. Using the same 500-HP transmission as an illustration, the power required to drive a closed-loop transmission system per gearbox is about 5 HP. With instrumentation accuracy of ± 1%, the observer can read power range from 4.95 HP to 5.05 HP. At 500 HP, this translated to 500 HP ± .05 HP. Therefore, if the sample transmission cited above is, in reality, 99% efficient, the observer can read the power loss within ± .05 HP versus ± 5 HP for an open-loop system.

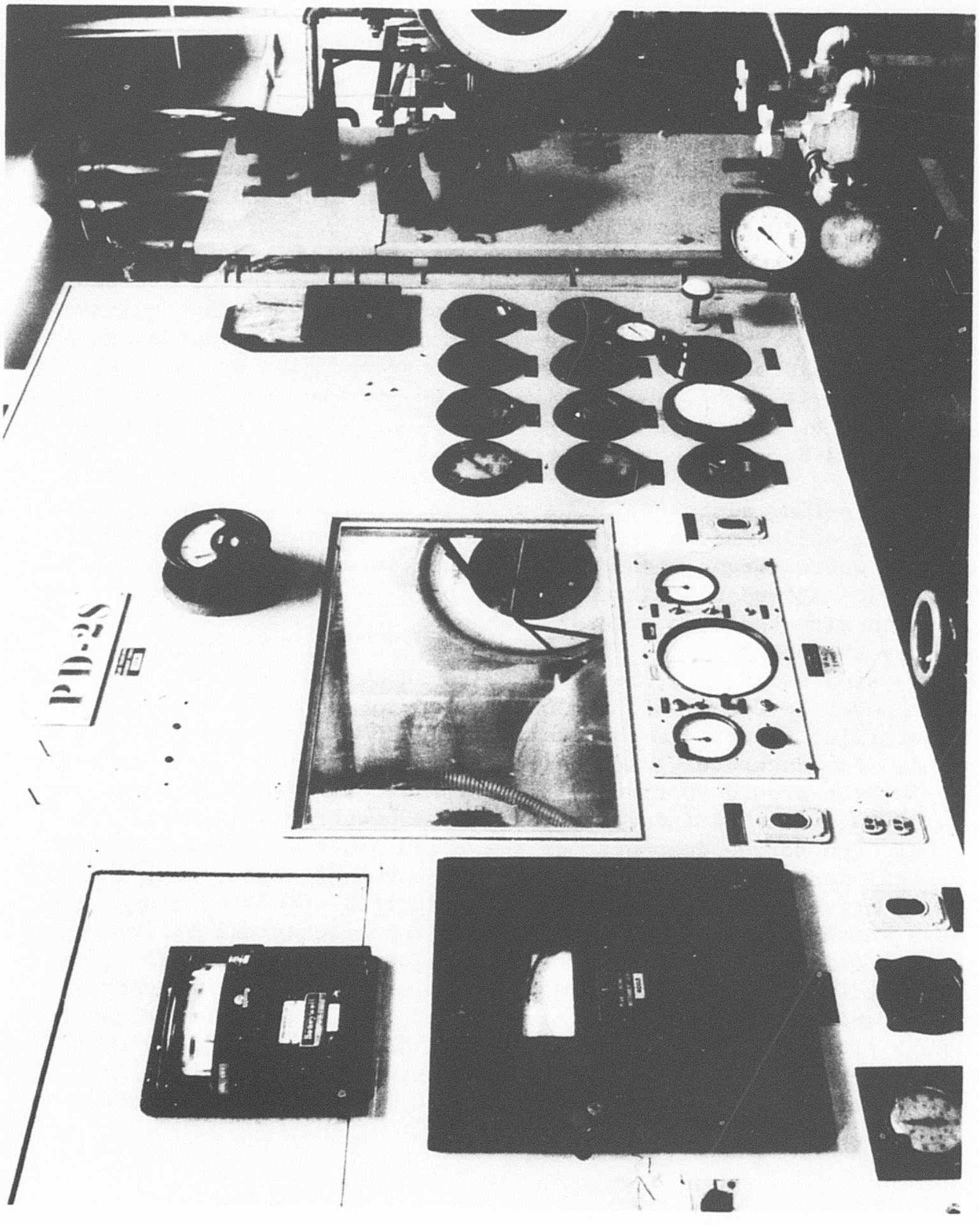


Figure 4. Control Panel.

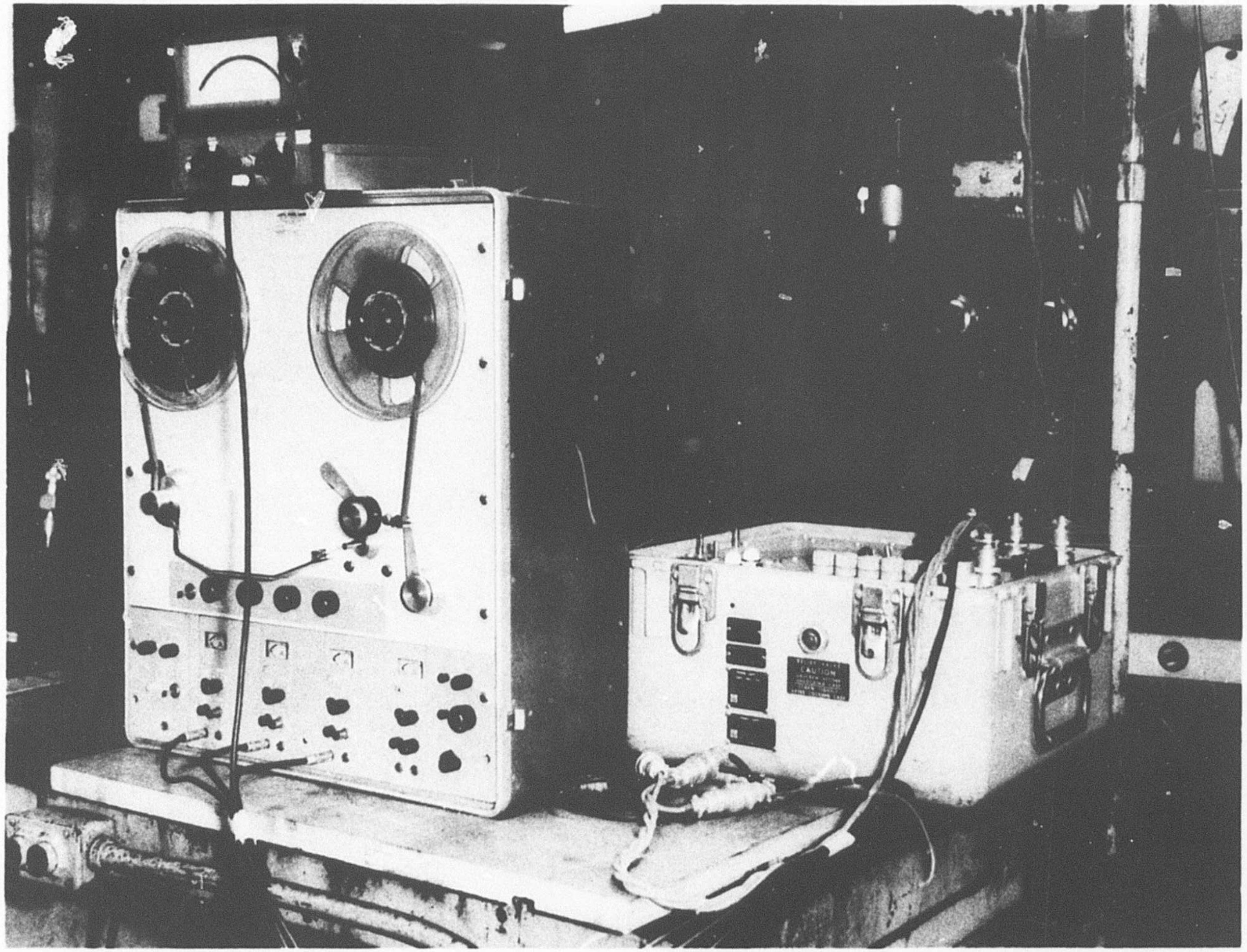


Figure 5. Setup for Recording Acoustic Emissions.

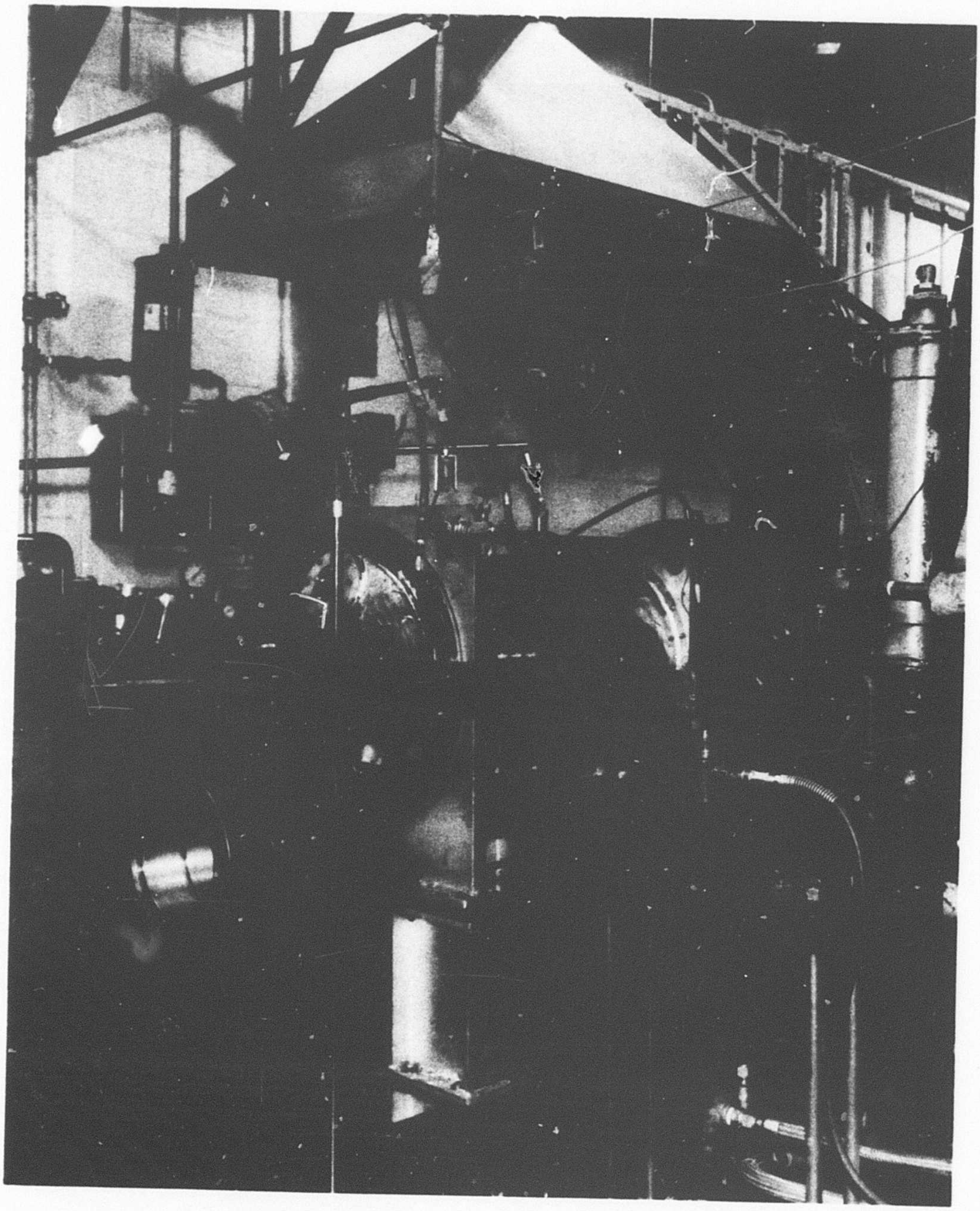


Figure 6. Test Rig on Test Stand.

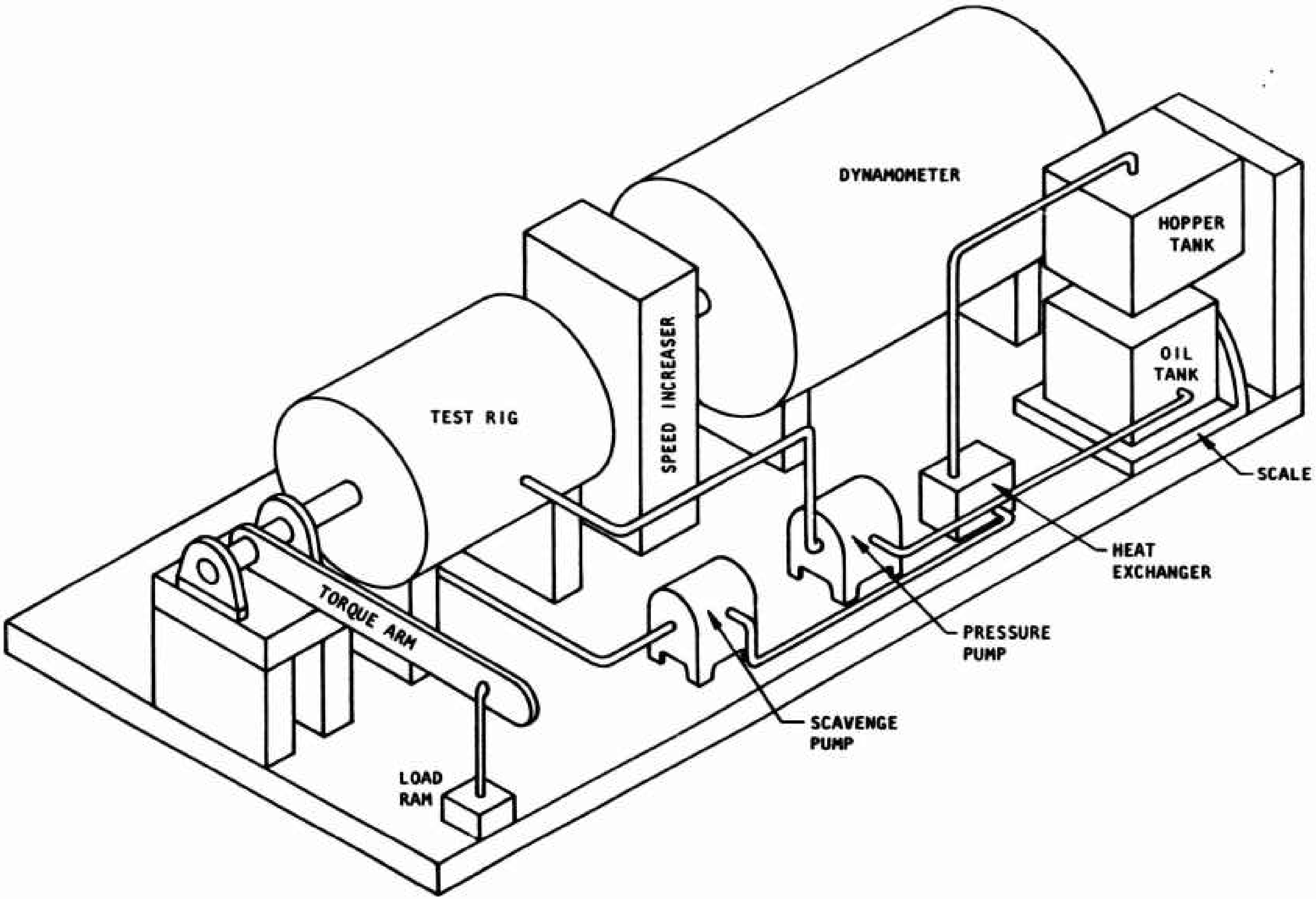


Figure 7. Free Planet Transmission Testing Arrangement.

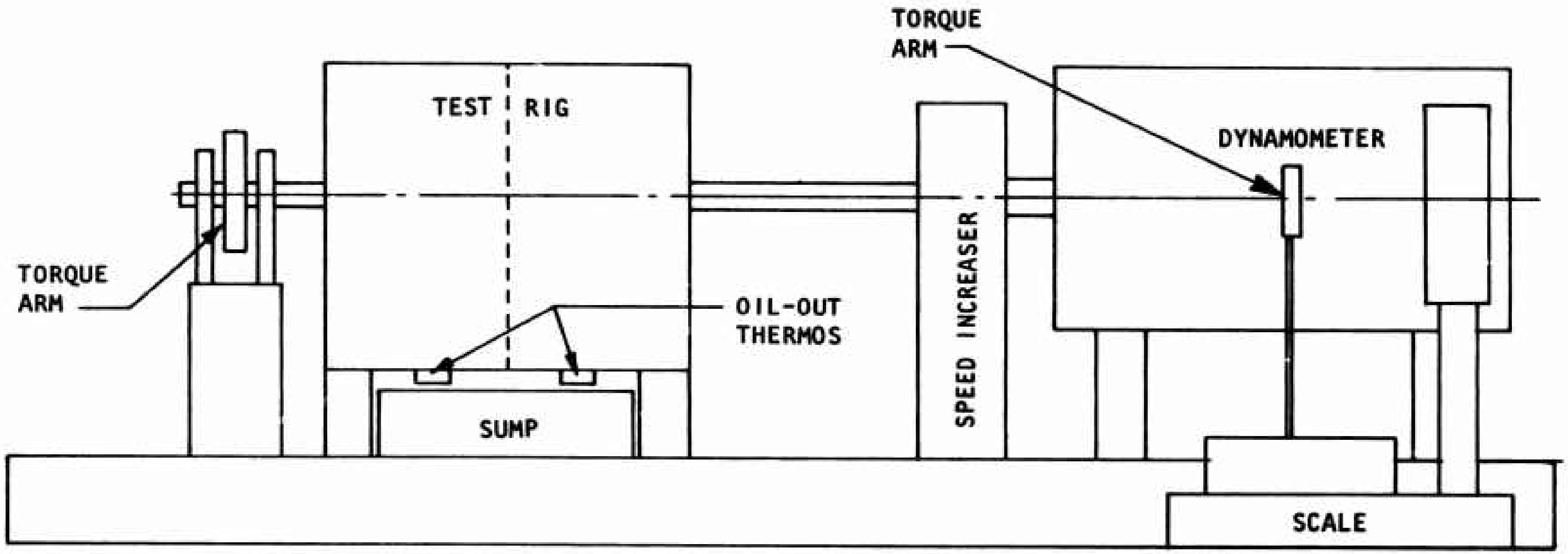


Figure 8. Free Planet Transmission Testing Arrangement.

TEST RESULTS

EVALUATION OF INCREASED BACKLASH

Initial feasibility demonstration of the free planet concept, while completely successful, did show a small degree of backloading on gear teeth of each mesh. In order to reduce or eliminate this condition, the three gear mesh backlash values were slightly increased by regrinding each gear on the planet spindle. The backlash was increased from the blueprint .003 to .005 inch by a nominal .002 inch to .005 to .007 inch. In addition to the increased backlash, the profile modifications were controlled to produce maximum blueprint tip and flank relief.

This modified configuration was subjected to 5 hours of operation at rated load and speed. At successful completion of this operation, the hardware was disassembled and inspected. Inspection of the antidrive side of the spindle gear teeth was made to determine if any indications of backloading were present. There were no markings or other indications of backloading at the fixed ring and sun meshes. The output mesh showed slight markings at one end of the tooth for about .125 inch. These findings contrast with essentially full face indications shown by the original free planet program.

During the 5-hour operation, the efficiency was determined. Results of this determination are shown in Figure 9. The reworks performed resulted in a slight increase in efficiency for this configuration. Table 1 shows representative efficiencies, calculated as shown in Appendix A, for random selected test data. Efficiencies calculated by measured torque and heat rejection are both shown. Figure 10 shows the hardware after the 5-hour shakedown run.

CYCLIC ENDURANCE

At the conclusion of the 5-hour break-in test and subsequent inspection, the units were rebuilt and a 50-hour cyclic endurance test was undertaken utilizing the duty cycle provided by Fort Eustis. This cycle is defined in the test program section of the report. This program was successfully completed, and inspection of the hardware after test showed no detrimental effects from this type of operation.

At approximately 16.75 hours into the cyclic endurance program, the C-W sonic monitoring system provided an indication of a change in noise signature of the "slave" gearbox. Upon disassembly and inspection of the test hardware, the test gearbox was found to be in satisfactory condition. The "slave" gearbox showed slight indications of gear tooth interference and slight rubbing contact of the sun planet mesh with the roller support ring face. Figure 11 shows this condition. This situation was corrected by increasing gear tooth break edge and reducing the width of the sun planet width by .015 inch. The cyclic endurance testing was then completed with the results noted above.

ACCELERATION AND DECELERATION

Acceleration testing was conducted as specified in the test program section of this report. Twenty-five cycles were conducted in which full speed (8000 rpm) was achieved in 10 seconds with a 10% applied load. Inspection of hardware after this portion of the program again showed no detrimental effect from this type of operation.

After the speed acceleration program, another 25 cycles utilizing the test points of the cyclic endurance program were conducted with load changes being accomplished in 2 seconds. Each test point was run for 5 minutes. Inspection of hardware after this evaluation showed no detrimental effect from this type of operation.

DYNAMIC EVALUATION

An investigation of spindle dynamics was conducted during the cyclic endurance phase of the program. This was accomplished utilizing a stroboscope to view the fixed ring mesh end of the planet spindles during operation. No instability of the spindle was observed during this investigation.

EFFICIENCY DETERMINATION

The efficiency was determined by direct reading of torque from the drive dynamometer and subtracting the premeasured losses from the test stand speed increaser and the calculated rig thrust bearing losses. This setup is shown schematically in Figure 8. Magnitude of efficiency was verified by calculating losses to the oil. Appendix A shows the methods used to calculate efficiencies. This includes both a mechanical and a thermal determination as shown in Table 1.

TEST VARIABLES

During all testing, the following test variables were controlled. A schematic of the lubrication system and thermocouple locations is shown in Figure 12.

1. Speed - as specified for the test point - 100% = 8000 rpm
2. Oil inlet temperature - 100 $\pm$ 5°F
3. Oil flow - constant 29 lb/min/gearbox and 11 lb/min to the rig thrust bearing
4. Torque loading - as specified for the test point, controlled by calibrated hydraulic pressure to the load ram

DATA RECORDED

1. Speed - measured by tachometer

2. Oil inlet temperature - thermocouple at entrance to the rig

   a. Test gearbox
   b. Slave gearbox
   c. Rig thrust bearing

3. Oil inlet pressure - pressure gauge at inlet to the rig

   a. Test gearbox
   b. Slave gearbox
   c. Rig thrust bearing

4. Load ram hydraulic pressure - pressure gauge

5. Dynamometer torque - by direct scale reading in pounds by a measured torque arm

6. Oil-out temperatures - thermocouples in drain lines

   a. Drain line test gearbox
   b. Drain line slave gearbox

7. Oil flows - controlled by calibrated pressure gauge at inlet to rig and shown schematically in Figure 12

   a. Test gearbox
   b. Slave gearbox
   c. Thrust bearing

   All temperature data was recorded by thermocouples to a Brown temperature recorder with a $\pm$ 1°F accuracy. Pressures were measured by gauges with a $\pm$ 0.1% accuracy.

8. Data measurement was accomplished with the following equipment.

   a. Speed-Tachometer - accuracy $\pm$ 2%
   b. Temperature - Brown recorder - accuracy $\pm$ 1°F
   c. Pressure - pressure gauges - accuracy 0.1%
   d. Load - platform scale - accuracy $\pm$ 1/4 lb

9. All measurement equipment was calibrated prior to test initiation and checked at the end of the program.

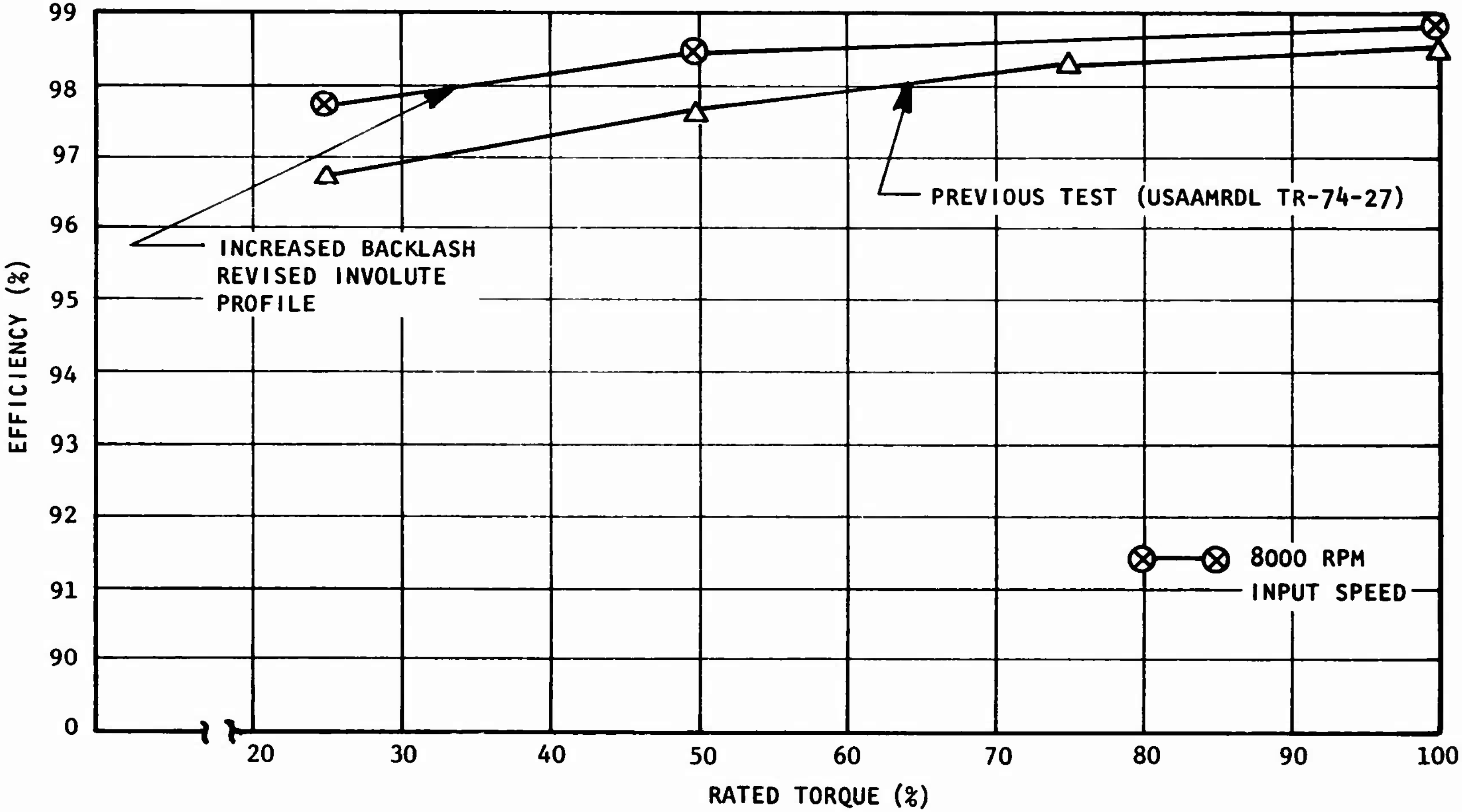


Figure 9. FP501R Efficiency During Cyclic Testing.

TABLE 1. FP501R TYPICAL EFFICIENCIES					
Condition	% Speed	% Load	Efficiencies		
			Mechanical	Thermal	
				Box 1	Box 2
1) 5-Hour Break-In	100	100	98.9	98.6	98.9
2) 50-Hour Cyclic Endurance	100	100	98.9	98.8	99.1
	100	25	97.9	96.3	97.3
	100	20	97.3	95.8	97.1
	100	100	98.9	98.8	99.1
	100	50	98.6	97.7	98.3
3) Cyclic Test-Rapid Load Change	100	100	99.0	99.0	99.3
	100	25	97.9	97.3	98.0
	100	20	97.3	96.7	97.5
	100	100	99.0	99.1	99.3
	100	50	98.6	98.1	98.7

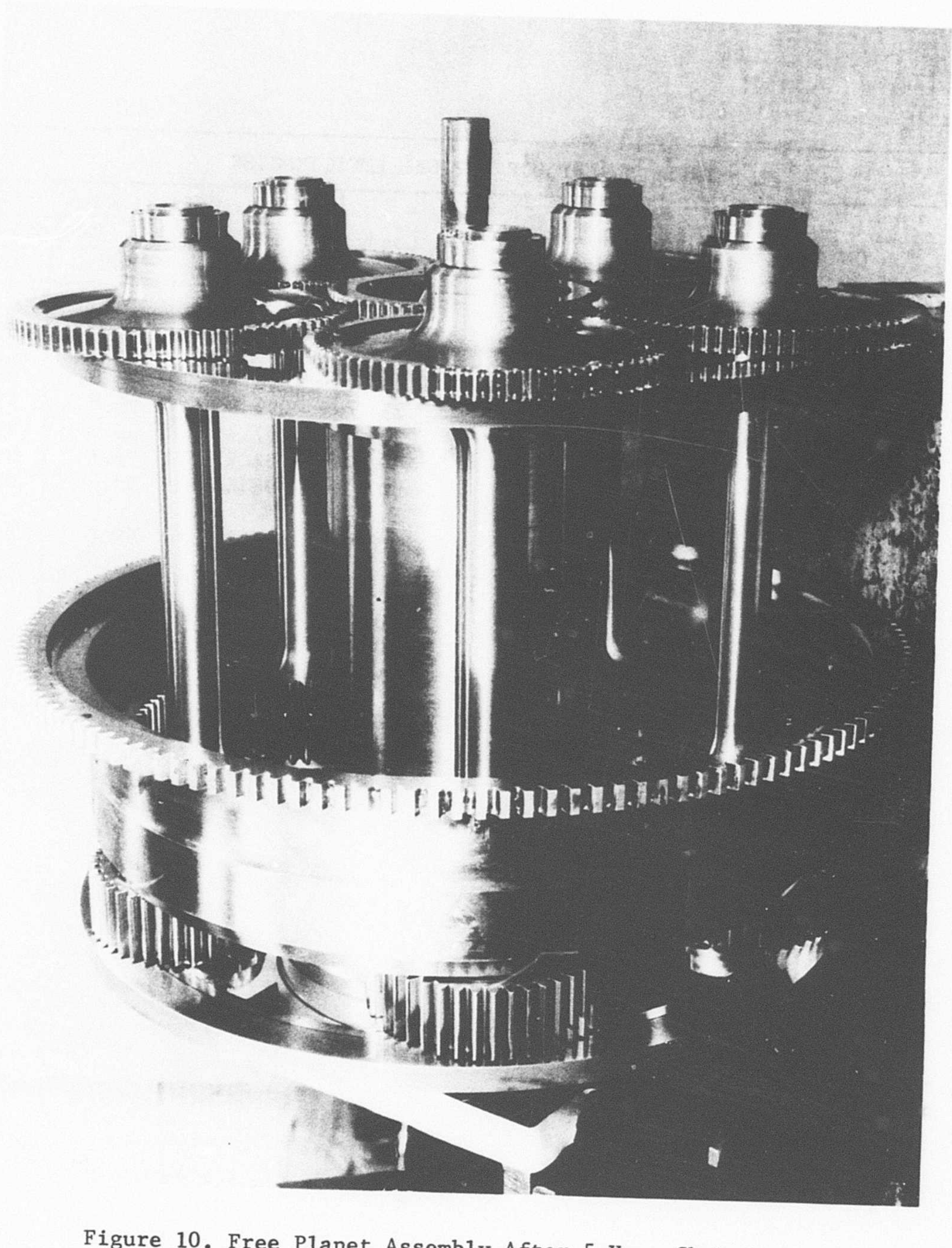


Figure 10. Free Planet Assembly After 5-Hour Shakedown.

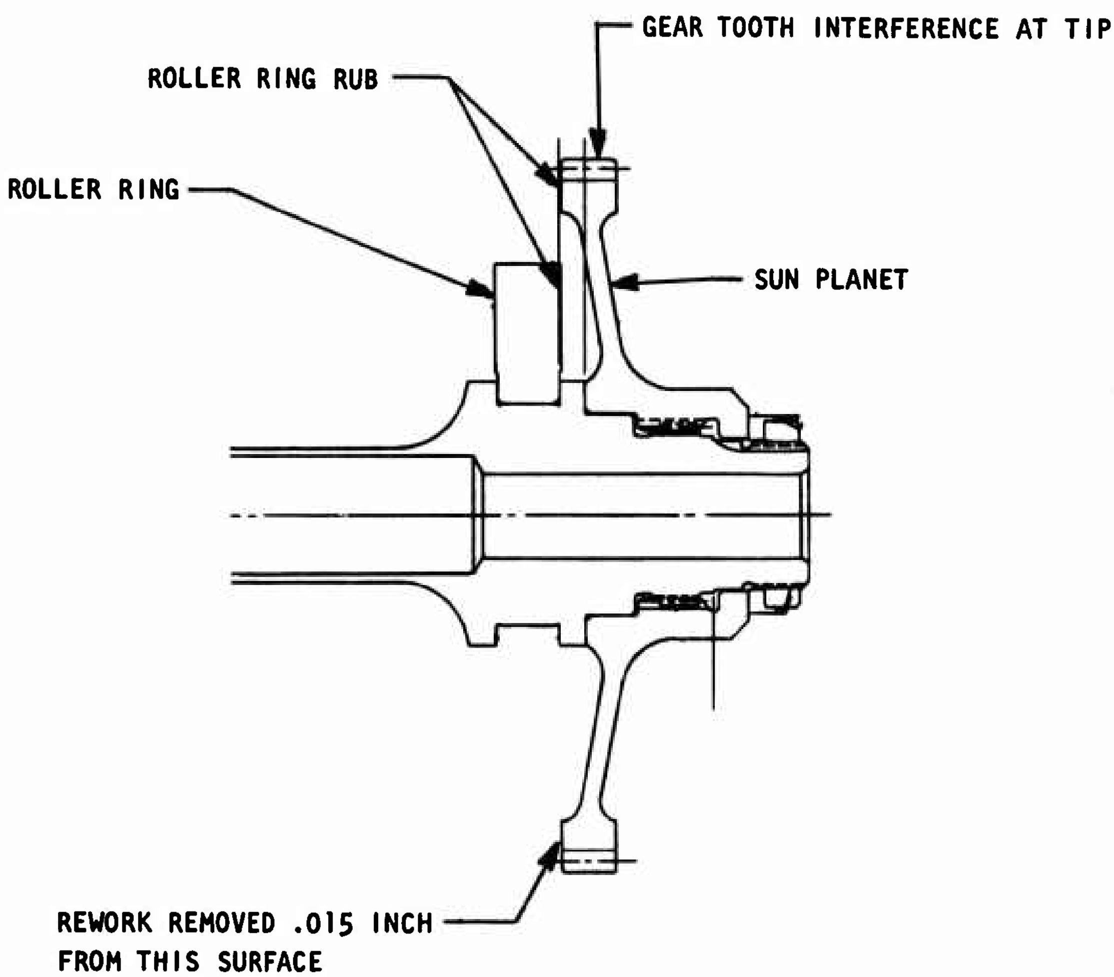


Figure 11. Slave Box Distress Area.

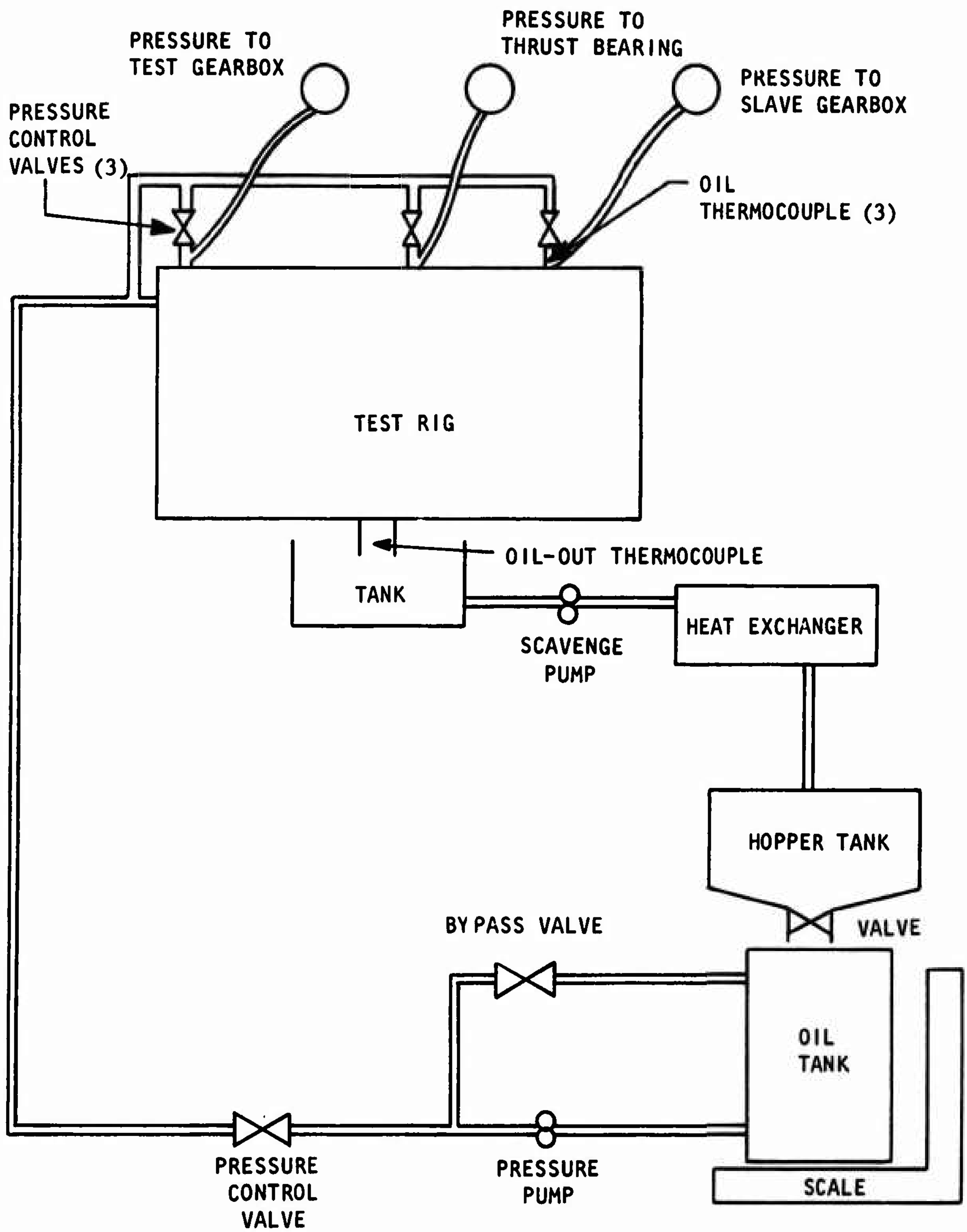


Figure 12. Free Planet Test Stand Lubrication and Instrumentation.

CONCLUSIONS

1. The force balance principle of the free planet transmission concept appears to be stable under both static and dynamic loading conditions.

2. The results achieved to date, including endurance testing and transient load testing, show that the free planet transmission concept is viable and that significant improvements over contemporary helicopter transmissions can be realized by its use.

3. Mechanical efficiency of 98.8% has been repeatedly recorded at full speed and rated torque on a 19.24 to 1 reduction ratio transmission. This high efficiency compares favorably with contemporary helicopter transmissions.

4. Good gear tooth pattern and load distribution among the spindles have been achieved on the prototype hardware.

5. Increasing the backlash to the upper limit of the blueprint specification improved the overall performance of the gear.

6. The dynamic investigation, including cyclic testing, acceleration evaluations and rapid load changing, showed the free planet transmission to be stable, with no detrimental effects.

7. On the basis of experimental work completed to date, it can be concluded with a high degree of confidence that the free planet transmission concept is stable and has high mechanical efficiency and good load distribution.

RECOMMENDATIONS

Based on the favorable results obtained to date from previous engineering studies, prototype hardware feasibility demonstration programs, and current development effort, it is recommended that the free planet transmission development program be expanded, and for the immediate future, the following areas be further explored:

1. Evaluate the effect of spindle tolerances on free planet transmission performances and cost considerations. The work just completed indicates that additional performance improvements might be possible and that at the same time manufacturing costs can be reduced even further.

2. Conduct extended durability evaluations of the free planet transmissions utilizing the existing hardware. The results of this type of effort will provide additional technology and guidance for the design and development of full-scale helicopter transmission systems.

APPENDIX A
EFFICIENCY CALCULATIONS

1. Mechanical efficiency

$$HP = \frac{TN}{63000} = \text{Dynamometer Horsepower}$$

T = (Scale reading, lb) (Moment arm, in.)

Moment arm = 22.24 in.

Scale reading is recorded for each test point.

N = Dynamometer RPM

Sample: $$HP = \frac{(52)\ (22.24)\ (1342)}{63000} = 24.6$$

Test stand speed increaser (Figure 8) has a measured power loss of 9.7 hp at rated speed.

The power loss in the test rig is the 24.6 - 9.7 = 14.9 hp.

Within the test rig is a large hydrodynamic thrust bearing to position the two test gearboxes. Its calculated loss is 4.7 hp.

The power loss of the two test rigs is then 14.9 - 4.7 = 10.2 hp.

Assuming the losses are evenly divided between the test gearbox and the slave gearbox, each has a 5.1-hp loss for a 500-hp power transmission.

$$\text{Efficiency} = \frac{500-5.1}{500} \times 100 = 98.98\%$$

2. Thermal efficiency

$$HP = \frac{(\text{Flow, lb/min})\ (\text{specific heat})\ (\Delta T)}{42.41}$$

Flow lb/min is measured

ΔT is measured

.5 = specific heat

$$HP = \frac{(34.5)\ (.5)\ (20)}{42.41} = 8.13$$

The total thrust bearing loss as in mechanical efficiency is 4.7 hp calculated. It is assumed the 50% is lost to each gearbox and that 50% of the flow goes to each gearbox. Horsepower loss is then 8.13 - 2.35 = 5.78.

$$\text{Efficiency} = \frac{500-5.78}{500} \times 100 = 98.84\%$$

APPENDIX B

REPRESENTATIVE EXPERIMENTAL TEST DATA

Condition	Torque Scale Rdg - Tare Pounds	Torque Scale Rdg - Pounds	Torque - Dynamometers In-Lb	% Torque Test Condition	Dynamometer RPM	Test Rig RPM Input	Oil Pressure psi Box 1	Oil Pressure psi Box 2	Oil Pressure psi Thrust Brg.	Oil Flow Lb/Min. Box 1	Oil Flow Lb/Min. Box 2	Oil Flow Lb/Min Thrust Brg.	Oil Inlet Temp. °F Box 1	Oil Inlet Temp. °F Box 2	Oil Inlet Temp. °F Thrust Brg.	Oil Drain Temp. °F Box 1	Oil Drain Temp. °F Box 2
1	28	80		100	1341	7994	20	20	11	29	29	11	104	103	104	127	123
2	28	80		100	1340	7988	20	20	11	29	29	11	100	100	100	120	117
	28	69.5		25	1339	7982	20	20	11	29	29	11	100	100	100	117	114
	28	69.5		20	1339	7982	20	20	11	29	29	11	98	98	98	114	111
	28	80		100	1340	7988	20	20	11	29	29	11	100	100	100	120	117
	28	73		50	1339	7982	20	20	11	29	29	11	97	97	97	117	113
3	28	79		100	1339	7982	20	20	11	29	29	11	99	98	99	117	114
	28	69.5		25	1341	7994	20	20	11	29	29	11	99	98	99	113	111
	28	69.5		20	1340	7988	20	20	11	29	29	11	98	97	98	112	110
	28	79		100	1340	7988	20	20	11	29	29	11	98	97	98	115	112
	28	73.5		50	1340	7988	20	20	11	29	29	11	98	98	98	115	112

Conditions
1) 5-hour break-in
2) 50-hour cyclic endurance
3) cyclic test rapid load change

6192-75